CYLINDRICAL ROLLER BEARING

BACKGROUND OF THE INVENTION

5   1. Field of the Invention

The present invention relates to a cylindrical roller bearing. In particular, it relates to a cylindrical roller bearing favorably incorporated in machine tools, engines such as jet engines or gas turbines, and other applications,

10 to support a high-speed rotating shaft.

2. Description of the Related Art

There are many cases where the main spindle assembly of a machine tool such as a machining center, a CNC lathe or milling machine, etc. is driven at high rotation for

15 reasons including improvement of work processing efficiency and precision. There is especially a marked recent trend toward increasing the main spindle rotation speed.

Generally, the main spindle of the main spindle assembly of a machine tool is supported to be freely

20 rotatable with respect to a housing by a rolling contact bearing. The rolling contact bearing is lubricated by oil mist lubrication, air / oil lubrication, jet lubrication, grease lubrication, or other such method in response to the conditions under which it is being used among other factors.

25 Cylindrical roller bearings, angular contact ball bearings,

etc. are used for rolling contact bearings.

Cylindrical roller bearings are generally composed of an inner ring having a raceway on an outer circumference thereof, an outer ring having a raceway on an inner

5   circumference thereof, a plurality of cylindrical rollers disposed to roll freely between the respective raceways of the inner ring and the outer ring, and a cage maintaining the cylindrical rollers spaced equally apart in the circumferential direction.

10   When flange portions are formed at both sides of the inner ring, recess grooves are respectively located at corner portions where flange surfaces of each of the flange portions of the inner ring and the raceway meet. The recess grooves are mainly formed as undercut grooves during

15   grinding of the raceways and the flange surfaces. Chamfers are located at angle portions where roller surfaces and both end surfaces of the cylindrical rollers meet respectively. Further, an axial dimension between the flange surfaces facing in the axial direction is set

20   slightly larger than the length of the cylindrical rollers, and thus a guide clearance is maintained between the cylindrical rollers and the flange portions.

Since the roller surfaces of the cylindrical rollers and the raceways of the races (inner and outer rings) are

25   in line contact, a cylindrical roller bearing such as has

2

been described has a high ability to handle radial load, and is suited for high-speed rotation. In contrast, when compared to a ball bearing, the amount of heat generated during high-speed rotation is large, and particularly, a

5   problem exists where increase in heat generated, wear at portions where there is sliding contact between the cylindrical rollers and the flange portions, and other effects occur easily.

Specifically, the cylindrical rollers have a degree

10   of freedom in inclination as large as the aforementioned guide clearance, and during rotation of the bearing, an unavoidable phenomenon occurs where the axes of the cylindrical rollers incline relative to the axis of the bearing, i.e., skew. When the cylindrical rollers skew, an

15   axial direction component is generated in driving force provided by the raceway on the revolving side, this becoming an axial thrust F to push end portions of the cylindrical rollers against one of the flange portions. Thus, there are instances of frictional resistance

20   increasing at regions where there is sliding contact between the cylindrical rollers and the flange portion, and the end portions of the cylindrical rollers and the flange portion thereby generate heat and suffer wear.

With respect to this problem, various methods of

25   improvement have been suggested in the related art. For

example, there are designs where the state of lubrication at the aforementioned portions of sliding contact is improved by designing the recess grooves with a larger height than the height of the chamfers of the cylindrical rollers and forming expanding tapered surfaces on the flange faces with a predetermined angle outward in the axial direction (for example, Japanese Patent Publication No. Sho 58-43609).

Also, there are designs that by having a structure wherein when the cylindrical rollers have skewed, by the outer circumferential edge portions of both end surfaces of the cylindrical rollers contacting with a portion nearer to the base end than the tip edges of the flange surfaces, the edge load of the portions of sliding contact is smaller compared to when the outer circumferential edge portions of both end surfaces of the cylindrical rollers contact with the tip edges of the flange surfaces (for example, Japanese Patent Laid-Open Publication No. Hei 7-12119).

As has been discussed, the cylindrical rollers have a degree of freedom in inclination as large as the guide clearance, and during rotation of the bearing, the cylindrical rollers rotate on their own axes as well as revolve about a shaft while their attitudes are ever-changing within a range of a maximum skew angle θ_{MAX}. The maximum skew angle θ_{MAX} here means a state where the outer

4

circumferential edge portions of both end surfaces of the
cylindrical rollers contact with the flange portions of
both sides of the raceway, being at the maximum skew angle
within the total degree of freedom of the skew angle of the
5   cylindrical rollers.

As shown in Fig. 10, when cylindrical rollers 23 skew
at a skew angle θ less than the maximum skew angle θ_{MAX},
the cylindrical rollers 23 are pushed in one direction
axially by the previously mentioned axial thrust F, and
10   roll while being guided in a state where they are pushed
against one of the flange portions 21b. In this case, the
state of contact between the cylindrical rollers 23 and one
of the flange portions 21b changes according to the skew
angle θ ($0 < \theta_T < \theta_U < \theta_{MAX}$) in the following manner.
15   In the range where the skew angle θ is $0 < \theta \leqq \theta_T$,
as shown in Fig. 11, a first boundary R_{13} between end
surfaces 23b and chamfers 23c of the cylindrical rollers 23
contacts a second boundary R_{11} between flange surfaces $21b_1$
and recess grooves 21c (contact point shown by a black
20   "●"). Then, in the range where the skew angle θ is $\theta_T <$
$\theta < \theta_U$, as shown in Fig. 12, the first boundary R_{13}
between the end surfaces 23b and the chamfers 23c of the
cylindrical rollers 23 contacts the flange surfaces $21b_1$
(contact point shown by a black " "). As the skew angle θ
25   next approaches θ_U, the first boundary R_{13} between the end

5

surfaces 23b and the chamfers 23c of the cylindrical

rollers 23 contacts with a third boundary R_{12} between the

flange surfaces $21b_1$ and flange chamfers $21b_3$ (not shown in

the drawings). After this, both end portions of the

5   cylindrical rollers 23 contact with both of the flange

portions 21b respectively to attain the maximum skew angle

θ_{MAX} (not shown in the drawings).

Fig. 13 shows the relation between the skew angle θ

of the cylindrical rollers 23 and the pressure P at the

10   contact surface of the cylindrical rollers 23 and the

flange portions 21b (solid line), and also shows the

relationship of the axial thrust F acting on the

cylindrical rollers 23 (broken line). As shown by the same

drawing, the axial thrust F becomes larger accompanying an

15   increase in the skew angle θ.

Within the range of $0 < \theta \leqq \theta_T$, a phenomenon occurs

where the pressure P at the contact surface increases at a

comparatively steep gradient accompanying an increase in

the skew angle θ. This is related to the cylindrical

20   rollers 23 and one of the flange portions 21b contacting at

the first boundary R_{13} and the second boundary R_{11} (the

state shown in Fig. 11), and the axial thrust F becoming

larger accompanying an increase in the skew angle θ.

Particularly, it has been verified through testing that the

25   pressure P at the contact surface becomes greater than a

6

surface pressure level P_0 where wear occurs at the regions

of contact thereof in the range $\theta_0 \leqq \theta \leqq \theta_T$ (region shown

by cross-hatching in the same drawing).

When the skew angle θ exceeds θ_T, the pressure P at

5   the contact surface decreases to a value below the surface

pressure level P_0 where wear occurs at the portions of

contact between the cylindrical rollers 23 and one of the

flange portions 21b, after which a stable transition is

shown at a comparatively low value irrespective of an

10 increase in the skew angle θ. This is related to a move

in the state of contact between the cylindrical rollers 23

and one of the flange portions 21b from contact between the

first boundary R_{13} and the second boundary R_{11} (the state

shown in Fig. 11) to contact between the first boundary R_{13}

15 and one of the flange surfaces $21b_1$ (the state shown in Fig.

12).

As the skew angle θ approaches θ_U, as shown in Fig.

13, the pressure P at the contact surface once again begins

a steep increase, and becomes a value exceeding the surface

20 pressure level P_0 at the point where θ_U has been reached.

This is related to the state of contact between the

cylindrical rollers 23 and one of the flange portions 21b

moving from contact between the first boundary R_{13} and the

flange surfaces $21b_1$ (the state shown in Fig. 12) to a

25 state of contact between the first boundary R_{13} and the

7

third boundary R_{12}.

As previously explained, the pressure P at the contact surface of the cylindrical rollers and the flange portions is at a value exceeding the surface pressure level P_0 where wear occurs at the portions of contact between the cylindrical rollers 23 and the flange portions 21b at the stage before reaching the maximum skew angle θ_{MAX}, that is, in the range where the skew angle θ is $\theta_0 \leqq \theta \leqq \theta_T$, $\theta_U \leqq \theta < \theta_{MAX}$. This is thought to be a major factor contributing to such problems as heat generation and wear at the contact portions.

However, the invention disclosed in Japanese Patent Publication No. Sho 58-43609 previously discussed makes no recognition of the aforementioned phenomenon, and does not suggest a way for improving it. Likewise, the invention disclosed in Japanese Patent Laid-Open Publication No. Hei 7-12119 restricts the state of contact between the outer circumferential edge portions of both end surfaces of the cylindrical rollers and the flange surfaces at the maximum skew angle θ_{MAX}, and there is no recognition of the discussed heat generation, wear, or other problems occurring at the stage $\theta_0 \leqq \theta \leqq \theta_T$, and does not suggest a means for improvement.

The discussed heat generation, wear, and other problems occurring at the stage $\theta_0 \leqq \theta \leqq \theta_T$ occur easily

8

in instances where a cylindrical roller bearing formed having flange portions only on one of the inner and outer rings (e.g., NU type, N type, etc.) is driven at high speeds with negative radial internal clearance (preload condition). Cylindrical roller bearings formed having flange portions only on one of the inner and outer rings do not handle axial load, and thus the skew of the cylindrical rollers occurs as a result of such factors as misalignment during bearing installation and slight imperfections in the shape of the raceways of the bearings. The skew angle thereof is therefore only slight with a high probability of being $\theta_0 \leqq \theta \leqq \theta_T$, and surface pressure at the contact portions between the cylindrical rollers and one of the flange portions becoming a value larger than the surface pressure level P_0 where wear occurs is a frequent state, which has been verified through testing. Further, heat generation, wear, etc. progresses easily when driven at high speeds, since the sliding speed at the portions where there is contact between the cylindrical rollers and the flange portions is high.

SUMMARY OF THE INVENTION

The present invention aims to improve a cylindrical roller bearing in light of the above knowledge.

Specifically, the task of the present invention is to

provide a cylindrical roller bearing suitable for higher

rotation speeds which controls heat generation, wear, and

other effects at the end portions of the cylindrical

rollers and the flange portions by decreasing the pressure

5   at the contact surface between the cylindrical rollers and

the flange portions, in particular the pressure at the

contact surface within a range where the skew angle of the

cylindrical rollers is $\theta_0 \leqq \theta \leqq \theta_T$.

A cylindrical roller bearing of the present invention

10   comprises an inner ring having a raceway on an outer

circumference thereof, an outer ring having a raceway on an

inner circumference thereof, a plurality of cylindrical

rollers disposed to roll freely between the raceway of the

inner ring and the raceway of the outer ring, flange

15   portions being disposed on both sides respectively of the

raceway of at least one of the inner ring and the outer

ring, and a recess groove disposed at a corner portion

where a flange surface of at least one of the flange

portions of both sides and the raceway meet, the flange

20   surfaces being inclined at the same angle from a base end

portion to a tip end portion thereof, a radial dimension h_3

of chamfers formed on outer circumferential edge portions

of the cylindrical rollers being set smaller than a radial

height h_1 from the raceway near the recess groove, curved

25   portions being formed between the chamfers and end surfaces

10

of the cylindrical rollers.

Curved portions, here, refer to the outer circumferential edge portions at the end surfaces of the cylindrical rollers being arc-shaped in cross section, the curvature radius of the arc thereof having a gradually changing shape along the radial direction. Further, in forming the curved portions, processing wherein a flexible hone contacts in a slightly inclined state with respect to the end surfaces of the cylindrical rollers is preferable.

The curved portions are regulated to be a shape wherein the following expressions are satisfied:

$$0.8 \leqq h_2/h_1$$

$$1 \leqq \tan^{-1} [\delta / (h_2 - h_3)] \, (°)$$

where h_1 is a radial height from the raceway near the recess groove, h_2 is a radial dimension from the roller surfaces of the cylindrical rollers to a boundary between the curved portions and the end surfaces, h_3 is a radial dimension of the chamfers of the cylindrical rollers, and δ is an axial dimension from a boundary between the chamfers and the curved portions to the end surfaces of the cylindrical rollers.

By the present invention possessing the aforementioned structure, problems occurring in the related art when the skew angle falls within the range $\theta_0 \leqq \theta \leqq \theta_T$, that is, the phenomenon of the boundary between the end

11

surfaces and the chamfers of the cylindrical rollers contacting with the boundary between the flange surfaces and the recess grooves, can be avoided. Therefore, generation of heat, wear, etc. at the end portions of the

5 cylindrical rollers and the flange portions can be prevented. A cylindrical roller bearing suited to higher rotation speeds can be provided, especially, a favorable cylindrical roller bearing for a main spindle assembly of a machine tool driven at higher rotation speeds.

10

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is an enlarged cross-sectional view showing essential portions of a cylindrical roller and a raceway of

15 an inner ring and a flange portion according to an embodiment of the present invention;

Fig. 2 is a cross-sectional view showing a structural example of a main spindle assembly of a machine tool;

Fig. 3 is a cross-sectional view including partially

20 omitted portions showing a cylindrical roller bearing according to an embodiment of the present invention;

Fig. 4 is a view showing a cylindrical roller and a flexible hone;

Fig. 5A is a view showing a cross-section when a

25 cylindrical roller is cross-sectioned on the plane A;

12

Fig. 5B is a view showing a state of contact between a cylindrical roller and a flange portion when a skew angle $\theta = 0°$ according to an example of the prior art;

Fig. 5C is a view showing a state of contact between the cylindrical roller and the flange portion when the skew angle is $0 < \theta \leqq \theta_T$ according to the example of the prior art;

Fig. 5D is a view showing a state of contact between a cylindrical roller and a flange portion when the skew angle $\theta = 0°$ according to an example of the present invention;

Fig. 5E is a view showing a state of contact between the cylindrical roller and the flange portion when the skew angle is $0 < \theta \leqq \theta_T$ according to the example of the present invention;

Fig. 6 is an enlarged view showing essential portions of a chamfer and a curved portion of a cylindrical roller according to an embodiment of the present invention;

Fig. 7 is a graph showing shape dimensions of curved portions of cylindrical rollers according to an embodiment of the present invention;

Fig. 8 is a characteristic graph of test results of embodiments showing the relationship between number of rotations and temperature increase;

Fig. 9 is a view of an end surface of a cylindrical

13

roller showing what are thought to be marks from slight

contact with a flange portion on a curved portion, obtained

as test results of an embodiment;

Fig. 10 is a view showing a state where cylindrical

5   rollers skew at a skew angle θ less than a maximum skew

angle θ_{MAX}, and are contacting with one of a pair of flange

portions;

Fig. 11 is a view showing a state where a boundary

between end surfaces and chamfers of cylindrical rollers

10   contacts with a boundary between a flange surface and a

recess groove;

Fig. 12 is a view showing a state where a boundary

between end surfaces and chamfers of cylindrical rollers

contacts with a flange surface; and

15   Fig. 13 is a graph showing a relationship between a

skew angle θ of cylindrical rollers of a cylindrical

roller bearing and a pressure P at the contact surface

between the cylindrical rollers and the flange portions

(solid line), as well as the relationship of an axial

20   thrust F acting on the cylindrical rollers (broken line),

according to the prior art.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

25   Fig. 2 shows a structural example of a main spindle

14

assembly of a machine tool such as a machining center or a grinding machine. The main spindle assembly shown in the same drawing is referred to as a built-in type, and is configured to rotatingly drive a main spindle 11 at high

5   speed by an internal motor 10. The motor 10 is disposed in the axial central portion of the main spindle assembly, and comprises a rotor 10a disposed on the outer circumference of the main spindle 11 and a stator 10b disposed on the inner circumference of a housing 12. When electric power

10 is supplied to the stator 10b, exciting power is generated between the stator 10b and the rotor 10a, and the main spindle 11 is rotatingly driven at high speed by the exciting power.

Rotation of the main spindle 11 is supported to be

15 freely rotatable relative to the housing 12 by rolling contact bearings disposed respectively at a front end (tool end) and a rear end (opposite to the tool end) sandwiching the motor 10. Normally, the rolling contact bearing at the rear end is structured to enable displacement in the axial

20 direction (free end), to absorb or let escape the amount by which the main spindle 11 axially expands due to heat during driving. In this example, matched mountings angular contact ball bearings (a pair of angular contact ball bearings) 13 are used as rolling contact bearings for the

25 front end, and a single-row cylindrical roller bearing 14

15

is used as a rolling contact bearing for the rear end.

Fig. 3 shows the cylindrical roller bearing 14 disposed at the rear end. The cylindrical roller bearing 14 comprises an inner ring 1 having a raceway 1a on the outer circumference thereof, an outer ring 2 having a raceway 2a on the inner circumference thereof, a plurality of cylindrical rollers 3 disposed to roll freely between the raceway 1a of the inner ring 1 and the raceway 2a of the outer ring 2, and a cage 4 holding the cylindrical rollers 3 spaced equally apart in the circumferential direction. Flange portions 1b are disposed on both sides respectively of the raceway 1a of the inner ring 1. The cage 4 can be formed from resin material.

The inner ring 1 and the outer ring 2 of the cylindrical roller bearing 14 are fitted to the outer circumference of the main spindle 11 and the inner circumference of the housing 12, respectively. A radial internal clearance during driving is for example a negative clearance. The internal parts of the bearing are lubricated by a method such as oil mist lubrication, air/oil lubrication, jet lubrication, or grease lubrication.

When the main spindle 11 is rotatingly driven at high speed by the motor 10 contained within the main spindle assembly, the rotation thereof is supported to be freely rotatable relative to the housing 12 by the angular contact

16

ball bearings 13 at the front end and the cylindrical

roller bearing 14 at the rear end. When the main spindle

11 undergoes expansion due to heat in the axial direction

owing to a temperature rise during driving, the amount of

5   expansion axially thereof is absorbed or allowed to escape

by sliding displacement between the outer ring 2 and the

cylindrical rollers 3 of the cylindrical roller bearing 14.

Further, when the present invention is applied to a

single-row cylindrical roller bearing, application is

10   possible to various known bearing types, including N type

(two inner-ring flanges), NU type (two outer-ring flanges),

NF type (two inner-ring flanges, one outer-ring flange), NJ

type (one inner-ring flange, two outer-ring flanges), or

NUP type (one of two inner-ring flanges is separable, two

15   outer-ring flanges). In an instance where the flange

portion is separable, undercut grooves are sometimes not

formed in the corner portions between the flange surfaces

and the raceways thereof. The present invention can also

be applied to double-row or multi-row cylindrical roller

20   bearings. In such instances as well, various known bearing

types can be adopted.

According to this embodiment, as shown in enlargement

in Fig. 1, recess grooves 1c are formed at the corner

portions where flange surfaces $1b_1$ of each of the flange

25   portions 1b and the raceway 1a of the inner ring 1 meet

17

respectively. The recess grooves 1c mainly are formed as

undercut grooves during grinding of the raceway 1a and the

flange surfaces $1b_1$. According to this embodiment, the

flange surfaces $1b_1$ are tapered surfaces inclined in a

5   direction so as to gradually open in a radially outward

direction, and flange chamfers $1b_3$ are formed at angle

portions where the flange surfaces $1b_1$ and outer radial

surfaces $1b_2$ of the flange portions 1b meet.

Chamfers 3c are formed at the angle portions where

10   roller surfaces 3a and both end surfaces 3b of the

cylindrical rollers 3 meet respectively. A first radial

dimension h_3 of the chamfers 3c formed on the outer

circumferential edge portions of the cylindrical rollers 3

is set smaller than a radial height h_1 from the raceway 1a

15   near the recess grooves 1c. Further, curved portions 3d

are formed between the chamfers 3c and the end surfaces 3b

of the cylindrical rollers 3.

The radial height h_1 is the dimension from the

location of the raceway 1a of the inner ring 1 to a first

20   boundary R_1 between the recess grooves 1c and the flange

surfaces $1b_1$. A second radial dimension h_2 is the dimension

from the location of a second boundary R_4 between the

roller surfaces 3a and the chamfers 3c to a third boundary

R_5 between the curved portions 3d and the end surfaces 3b

25   of the cylindrical rollers 3. The first radial dimension

18

h_3 is the dimension from the location of the second

boundary R_4 between the roller surfaces 3a and the chamfers

3c to a fourth boundary R_3 between the chamfers 3c and the

curved portions 3d. Further, it is preferable that

5   boundaries R_1 through R_5 be formed with curved surfaces to

decrease pressure at the contact surface, for example, with

arcuate surfaces having a curvature radius of 0.1 to 0.3 mm,

to form a smooth continuation with adjacent surfaces.

The reason for setting the first radial dimension h_3

10   of the chamfers 3c of the cylindrical rollers 3 smaller

than the radial height h_1 from the raceway 1a near the

recess grooves 1c will now be explained.

For example, if the first radial dimension h_3 of the

chamfers 3c of the cylindrical rollers 3 is set larger than

15   the radial height h_1 from the raceway 1a near the recess

grooves 1c, as is the case with the cylindrical rollers 23

(see Fig. 12), the first boundary R_{13} between the end

surfaces 23b and the chamfers 23c thereof contacts with the

flange surfaces $21b_1$ in a state where the cylindrical

20   rollers 23 are not skewing. Normally, since the chamfers

23c are processed by forging, precision in the form of the

first boundary R_{13} is poor. It has been verified through

testing that in a state where the cylindrical rollers 23

are not skewing, abnormal vibration occurs in the

25   cylindrical rollers 23 when the first boundary R_{13} with

19

poor form precision contacts with the flange surfaces $21b_1$, and wear occurs at the first boundary R_{13}. Therefore, according to this embodiment, the first radial dimension h_3 of the chamfers 3c of the cylindrical rollers 3 is set

5   smaller than the radial height h_1 from the raceway 1a near the recess grooves 1c.

The previously mentioned curved portions 3d are regulated to satisfy the following conditions:

$$0.8 \leqq h_2/h_1$$

10     $$1 \leqq \tan^{-1} [\delta / (h_2 - h_3)] \; (°)$$

where h_1 is the radial height from the raceway 1a near the recess grooves 1c, h_2 is the second radial dimension from the roller surfaces 3a of the cylindrical roller 3 to the third boundary R_5 between the curved portions 3d and the

15   end surfaces 3b, h_3 is the first radial dimension of the chamfers 3c of the cylindrical rollers 3, and δ is the axial dimension from the fourth boundary R_3 between the chamfers 3c and the curved portions 3d to the end surfaces 3b of the cylindrical rollers 3.

20   Further, as shown in Fig. 4, the curved portions 3d can be made through processing wherein a flexible hone 5 is brought into contact therewith in an inclined state of only a slight angle α relative to the end surfaces 3b of the cylindrical rollers 3.

25   By forming the curved portions 3d between the

20

chamfers 3c and the end surfaces 3b of the cylindrical

rollers 3, the problems of the prior art which occur when

the skew angle θ falls within the range $0 < \theta \leqq \theta_T$

(including $\theta_0 \leqq \theta$), that is, the phenomenon of the first

5   boundary R_{13} between the end surfaces 23b and the chamfers

23c of the cylindrical rollers 23 contacting with the

second boundary R_{11} between the flange surfaces $21b_1$ and the

recess grooves 21c (see Fig. 11) can be avoided. This will

now be explained referring to Figs. 5A to 5E.

10   Figs. 5B and 5C, as shown in Fig. 5A, show a cross

section A of the cylindrical rollers 23 and the second

boundary R_{11} of an example of the prior art observed from a

direction radially outside the bearing toward the bearing

center. Figs. 5D and 5E, as shown in Fig. 5A, show a cross

15   section A of the cylindrical rollers 3 and the first

boundary R_1 of the present invention observed from a

direction radially outside the bearing toward the bearing

center.

According to the example of the related art, as shown

20   in Figs. 5B and 5C, when the cylindrical rollers 23 skew in

the range $0 < \theta \leqq \theta_T$ (including $\theta_0 \leqq \theta$), the first

boundary R_{13} between the end surfaces 23b and the chamfers

23c thereof contacts with the second boundary R_{11} between

the flange surfaces $21b_1$ and the recess grooves 21c. In

25   this instance, since both the edges of the cylindrical

21

rollers 23 and the flange surfaces 21b are contacting, the

pressure at the contact surface is large, with abnormal

heat generation, wear, etc. occurring.

In contrast, according to the example of the present

5   invention, as shown in Figs. 5D and 5E, when the

cylindrical rollers 3 skew within the range $0 < \theta \leqq \theta_T$

(including $\theta_0 \leqq \theta$), the curved portions 3d contact with

the first boundary R_1 between the flange surfaces $1b_1$ and

the recess grooves 1c. In this instance, pressure at the

10   contact surface is small since mutual contact between edge

portions as with the example of the related art can be

avoided, and abnormal heat generation, wear, etc. do not

occur.

The reason for setting the conditions for forming the

15   curved portions 3d as $0.8 \leqq h_2/h_1,$ $1 \leqq \tan^{-1} [\delta / (h_2 -$

$h_3)]°$ is as follows (h_1: radial height from the raceway 1a

near the recess grooves 1c; h_2: second radial dimension

from the raceway 3a to the third boundary R_5 between the

curved portions 3d and the end surfaces 3b of the

20   cylindrical rollers 3; h_3: first radial dimension of the

chamfers 3c of the cylindrical rollers 3; δ : axial

dimension from the fourth boundary R_3 between the chamfers

3c and the curved portions 3d to the end surfaces 3b of the

cylindrical rollers 3).

25   Normally, the first radial dimension h_3 of the

22

cylindrical rollers 3 is in a relationship with the radial height h_1 of about $0.5 \leqq h_3/h_1 \leqq 0.9$, and even in an instance of $h_3/h_1 = 0.5$, the lower limit value of h_2/h_1 is 0.8 so as to maintain the radial width $(h_2 - h_3)$ of the curved portions 3d of the cylindrical rollers 3.

Also, normally for cylindrical rollers 3, even at the maximum skew angle θ_{MAX} it is only an angle of approximately 0.5 to $1.0°$, however, $1 \leqq \tan^{-1} [\delta / (h_2 - h_3)]°$ is set such that the curved portions 3d will contact with the first boundary R_1 between the flange surfaces $1b_1$ and the recess grooves 1c when the cylindrical rollers 3 skew within a range of $0 < \theta \leqq \theta_T$ (including $\theta_0 \leqq \theta$) when an apex angle θ_1 of a triangle formed by the radial width $(h_2 - h_3)$ of the curved portions 3d of the cylindrical rollers 3 and the axial dimension δ (see Fig. 6) is $1°$ or greater. Verification was carried out with regard to this by the following test using manufactured products where $\tan^{-1} [\delta / (h_2 - h_3)] = 1°$ is satisfied.

(Embodiments)

Test bearings for the examples of the prior art and the present invention were fabricated and were placed in a test apparatus. Driving operation was carried out under the hereinafter described conditions, wherein the relationship between number of revolutions and temperature increase was sought after, and the conditions of wear

23

occurrence at the end portions of the cylindrical rollers was observed.

(Test conditions)

Test bearings: single-row cylindrical roller bearing

5 N1020 (prior art example, present invention example)

Number of revolutions: 2000 to 8000 rpm

(maximum dn = 80×10^4)

internal clearance: $- 5\mu$m (radial internal clearance after assembly)

10 Lubrication: grease lubrication

Shape of curved portions 3d (see Fig. 7): $h_2/h_1 = 1.05$

$\tan^{-1} [\delta / (h_2 - h_3)] = 1°$

(Test results)

15 Results of temperature increase with the example of the prior art and the example of the present invention are shown in Fig. 8. With the example of the prior art, temperature increase of the bearing is high, and also, the bearing temperature became unstable at 7000 rpm, and

20 therefore the test could not be continued. Upon observing the bearing following the test, the occurrence of wear was verified at the first boundary R_{13} between the end surfaces 23b and the chamfers 23c of the cylindrical rollers 23, and at the second boundary R_{11} between the flange surfaces 21b$_1$

25 and the recess grooves 21c of the inner ring 21.

In contrast, with the example of the present

24

invention, temperature increase of the bearing was low, and
stable driving at 8000 rpm was possible. Driving operation
was thus carried out for 1000 hours, after which the
bearing was observed. Wear was not visible at the end

5   surfaces 3b of the cylindrical rollers 3, with only what
are thought to be contact marks a on the curved portions 3d
resulting from slight contact with the flange portions 1b
of the inner ring 1 (see Fig. 9), and thus effects of the
present invention were verifiable. Moreover, effectiveness

10   of the present invention when $\tan^{-1} [\delta / (h_2 - h_3)] = 1°$ was
satisfied was also verified. Therefore, the lower limit
value of $\tan^{-1} [\delta / (h_2 - h_3)]$ is set as 1.